Advanced-Cycle Power Systems Utilizing Desulfurized Fuels

F. L. Robson
A. J. Giramonti

United Aircraft Research Laboratories
East Hartford, Connecticut

INTRODUCTION

The electric utility industry in the United States currently contributes approximately 50% of the nearly 30 million tons per year of sulfur oxides emitted into the atmosphere (Ref. 1). Since the total installed capacity of electric utilities is projected to double each decade, (Ref. 2) the amount of sulfur oxides emitted into the atmosphere in future years could exceed projected standards in some sections of the country unless suitable methods are developed to control sulfur oxide emissions. Many processes, ranging from cleanup of the stack gas to cleanup of fuel before combustion, have been proposed for controlling sulfur oxide emissions from power plants. Although many stack gas cleaning methods are technically feasible, most of them are expensive and have not demonstrated reliable operation in commercial service. The alternative approach, involving removal of sulfur from fuel before combustion, looks most promising as a long-range solution for controlling sulfur oxide pollution from fossil-fueled central power stations (Ref. 3).

The removal of sulfur from fossil fuels before combustion can be a difficult task, and the resulting fuel delivered to the power system is certain to cost more than the raw fuel which serves as feedstock. In order to evaluate the technical and economic feasibility of fuel desulfurization processes as an alternative to stack gas desulfurization, it is necessary to reappraise the traditional methods of electric power generation and to evaluate advanced power systems which may be capable of operating at higher efficiencies than conventional steam systems. The Research Laboratories of United Aircraft Corporation, under contract to the National Air Pollution Control Administration*, are currently investigating the technical and economic feasibility of desulfurizing coal and residual oil and the utilization of these desulfurized fuels in advanced-cycle power systems. This paper describes the results of cycle analysis of various advanced power systems including preliminary estimates for the cost of generating power with these systems and indicates the benefits that may arise through the use of desulfurized fuels.

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FUEL PROCESSING

While this paper deals exclusively with the preliminary results of a study on advanced-cycle power systems, it is appropriate to discuss briefly at this time the type of fuel used in this investigation. In order to meet sulfur oxide emission control standards, the sulfur contained in coal or residual oil must be significantly reduced or removed altogether. One method of obtaining a low-sulfur fuel is through the partial oxidation of coal with steam and air at high temperature and pressure. The product gas from this process, with sulfur now in the form of H_2S and COS is then sent to a desulfurization unit which removes the sulfur compounds. Since the gasification processes and desulfurization techniques of interest would typically operate at elevated pressures, a high-pressure fuel gas having a heating value of 150 to 200 Btu/ft^3 at standard conditions would result. Such a fuel gas has a number of advantages in combination with the power systems to be discussed, and therefore this type of fuel was assumed in all the systems investigated.

Preliminary estimates of the cost of potentially attractive gasification processes have indicated that a clean, desulfurized fuel could be delivered to a mine-mouth powerplant for a cost which is 25% to 50% greater than the cost of the raw fuel.

ADVANCED-CYCLE POWER SYSTEMS

The specific types of advanced-cycle power systems investigated are shown in Table I, and can be grouped into two generic classifications: external-combustion power systems in which boilers or heaters are used to heat the working fluid, and internal-combustion power systems in which the products of combustion constitute the working fluid. The external-combustion systems investigated include those using the conventional steam cycle, binary cycles utilizing steam and other working fluids, and cycles such as the closed-cycle gas turbine in which the working fluid is heated in a gas heater. The internal-combustion systems studied were essentially based on variations of the gas turbine cycle and included consideration of power systems using the combined gas and steam (COGAS) turbine cycles. The investigations were based on present-day power system technology, although possibly not yet reduced to commercial practice, as well as technology judged to become commercially available in the 1980 and 1990 decades.

Steam Systems

Conventional steam power systems were included in the investigation to provide a basis of comparison for all other power systems. Performance estimates predicted for projected future conventional steam power systems are given in Table II. It is apparent from this table that the overall station efficiencies for these systems are not projected to increase substantially in the time span considered, because technology available for use in steam power stations has reached a plateau. Relatively minor increases in station efficiency will be possible due to slight increases in the internal efficiencies of turbogenerators and boilers, but substantial increases in efficiency due to improved cycle conditions are not foreseen, since any increase in cycle temperature and pressure would result in a very marked increase in system capital costs. Thus, no significant increase in conventional steam power station thermal efficiency is foreseen to offset the increased costs projected for future desulfurized fuels.

Binary Cycles

The basic steam power station may be modified by the addition of a binary cycle to either increase its efficiency or decrease its capital cost, both with the goal of maintaining or reducing the cost of power. One method which has been suggested (see Ref. 4, for example) for reducing the capital cost of power stations is to stop the steam expansion at approximately 35 psia, eliminate the relatively expensive low-pressure sections of the steam turbine, and incorporate an ammonia or fluorocarbon bottoming cycle which would operate at relatively high pressure. Supposedly, the capital cost of the bottoming system would be less than for the steam equipment it replaces. A temperature-entropy diagram for a 3500 psig/1000 F/1000 F steam cycle with an ammonia bottoming cycle is depicted in Fig. 1. The efficiency of a power station incorporating this type of a bottoming cycle would be substantially less than that of a 3500 psig/1000 F/1000 F steam station, i.e., approximately 35.7% compared to 38.6%, because of the irreversible heat transfer between steam and ammonia, and because it is estimated that the ammonia turbine would be capable of attaining a slightly lower isentropic efficiency than the section of the steam turbine which it would replace. Analysis has shown that the increased fuel consumption due to reduced cycle efficiency would more than offset any system capital cost reductions that might be anticipated with a bottoming cycle. In fact, when account is taken of various capital cost penalties associated with the use of fluids other than steam, such as the need to employ welded construction to minimize leakage, the total capital cost of a bottoming cycle would not be significantly different than the cost of the conventional steam equipment it would replace.

A method which has been suggested for increasing the performance of the power stations is to use a high-temperature topping cycle which would reject its heat in a steam boiler. Previous studies (Refs. 5 and 6) of this type of cycle have indicated that potassium would be the best topping-cycle working fluid. In order to eliminate the upper limit of system performance, the potassium topping cycle shown in Fig. 2 was analyzed. The potassium would be pumped to its highest pressure of only 152 psia, heated to 2000 F by combustion gases, expanded through a turbine equipped with moisture separators to keep the moisture content of the potassium from exceeding 12%, and finally exhausted into a steam heater wherein the potassium would be condensed. If such a potassium cycle were used in conjunction with a 3500 psig/1000 F/1000 F steam cycle, a binary cycle efficiency of 58.8% would result. The overall station efficiency, taking into account such factors as boiler losses, generator efficiencies, and pump losses, would be 50.6%. This value is more than 10 points higher than the efficiency of the best present-day steam plant. As in the case of the bottoming cycles, the cost of the system equipment would be penalized in comparison with conventional steam systems because of the necessity to eliminate leakage of the working fluid and also to provide safety equipment to minimize the potential effects of a potassium-water reaction. Also, the heat exchangers require very costly materials for construction. Thus, the estimated capital cost for this system of over $200/kw is significantly higher than that for conventional plants, and it is estimated that the total cost of producing electricity would not be reduced relative to the cost with conventional steam stations. Furthermore, the problems associated with the successful development of potassium turbines of several hundred megawatts capacity are very complex.

Closed-Cycle Systems

A second group of external-combustion cycles is formed by what could be called closed-cycle gas turbine systems. The systems investigated included a true Brayton-cycle system utilizing helium as the working fluid, a supercritical Rankine-cycle system utilizing CO_2 as the working fluid, and a combination Rankine and Brayton-cycle system utilizing SO_2 as the working fluid.

Several cycle configurations involving the use of intercooling, regeneration, and reheating were studied. Considerations of advanced materials suitable for use in the fluid heaters indicated that tube wall temperatures would have to be restricted to 1800 F or below if an acceptable equipment lifetime of at least 100,000 hr were to be obtained. Thus, it was decided that the maximum working fluid temperature would be limited to 1600 F, and cycle evaluations and equipment sizing were performed for this value. A second temperature level of 1200 F at which advanced, but currently available, materials could be used was also selected for evaluation. By investigating a number of configurations for these two temperature levels, the tradeoff between cycle efficiency and equipment capital cost could be estimated.

Helium closed-cycle gas turbine systems have been the subject of widespread interest (e.g., Refs. 7, 8, and 9) because of the potentially high cycle thermal efficiencies such as those shown in Fig. 3. The efficiencies shown are for a cycle having one intercooler and a 90% effective regenerator. For the 1600-F temperature limit, the efficiency would be approximately 47%, and at the 1200-F level, approximately 44%. This efficiency can be increased somewhat by going to a different configuration, and the system selected for further evaluation at 1600 F would use two intercoolers and a 90% effective regenerator to give an estimated 48.5% cycle efficiency. The resulting power station would have a net efficiency of 41% with an estimated installed cost of $170/kw.

The use of CO_2 as a working fluid has been investigated a number of times, Refs. 9 and 10 being prime examples. Because of its low critical temperature, 88.7 F, CO_2 cannot be used in a Rankine cycle since the minimum cycle temperature allowed by the available cooling water is approximately 100 F. A typical cycle using CO_2 is shown in Fig. 4 in which it is seen that the flow would be split into two streams, one being compressed in a gas compressor and the other being cooled to supercritical liquid and then pumped up to maximum cycle pressure. A configuration such as this reduces the total compressor work required and would also allow the use of extensive regneration. The configuration of Fig. 4 would result in an overall station efficiency of 39% at an estimated capital cost of about $200/kw.

The final working fluid considered for the external-combustion cycles was SO_2. While this fluid is toxic, it does exhibit other properties which make it an interesting fluid for power systems (Ref. 11). The critical temperature of SO_2 is 315 F; thus, a condensing cycle can be considered. The 1600-F cycle selected for evaluation is shown in Fig. 5. Like the CO_2 cycle, the flow would be split into two streams: one compressed as a gas, the second condensed to the liquid state and pumped to maximum cycle pressure. By utilizing a reheat cycle and a 92.5% effective regenerator, this cycle would exhibit an efficiency of nearly 59%. The overall station efficiency would be about 51% with an installed cost estimated to be $227/kw, an appreciable portion of which can be attributed to the very large regenerator.

Thus, a number of variations of external-combustion cycles have been investigated with the intent of increasing efficiencies or decreasing capital cost in order to offset the potential increase in fuel cost which would result from using desulfurized fuel. The potential power costs for systems using these cycles are summarized in Fig. 6 in which the estimated generating cost in mills per kilowatt hour for each system is compared with that of a conventional steam system. The generating costs are given for two fuel costs, 30¢/million Btu, which is a typical price of present-day untreated residual fuel oil, and 50¢/million Btu, a price projected for typical future desulfurized fuels. The results presented in Fig. 6 show that none of the cycles discussed thus far demonstrate cost advantage over the conventional steam system.

Gas Turbine Systems

The second generic group of power systems considered for use in central stations consists of internal combustion systems in which the products of combustion constitute the working fluid. Contrary to the case for conventional steam systems in which no significant improvement in performance is foreseen during the time period of interest, industrial gas turbine technology is projected to continually improve during the next several decades, primarily because of fallout from advanced aircraft development programs (Refs. 12, 13, and 14). The use of aircraft gas turbine technology in large industrial-type gas turbines could then give rise to performance benefits that would allow these engines to become competitive with steam power systems.

Figure 7 lists some aspects of gas turbine technology for the three time periods under consideration. The projections shown in Fig. 7 indicate that both aerodynamic performance (i.e., compressor and turbine efficiencies) and turbine inlet temperatures increase with time. The projected improvements in turbine inlet temperature are due to two considerations: increases in materials technology, which allow blade materials to withstand higher operating temperatures, and improvements in blade cooling techniques. Historically, turbine inlet temperatures have advanced approximately 20 F per year because of materials improvements. This improvement is shown in Fig. 8 along with the improvement made possible by the use of several cooling techniques. Data points in Fig. 8 indicate actual or projected engines utilizing both improved materials and improved cooling techniques.

A major improvement in gas turbine performance could be realized if the compressor bleed air normally used to cool the turbine blades is precooled in an external heat exchanger to temperatures of about 125 F before being utilized in the turbine for cooling purposes. The performance improvements that would result from the use of precooled compressor bleed air are: (1) for the same amount of bleed air extraction, a higher turbine inlet temperature could be realized, or (2) a smaller extraction flow would be required to maintain a given turbine inlet temperature. The performance gains which might then be realized by using precooled bleed air are shown in Fig. 9, in which projected performance for three generations of engines is shown. Another benefit which might arise from the use of precooled compressor bleed air is that less costly impingement cooling might be used instead of transpiration cooling in very high-temperature engines.

The performance shown in Fig. 9 was based on the use of methane as fuel. The use of a fuel resulting from gasification of coal would actually improve the performance over that of an engine burning methane. This improvement is shown in Fig. 10 for an advanced-design engine. The improved performance results because the fuel gas supplied from a high-pressure (above 15 atm) coal gasification facility typically would have a low heating value (150 to 200 Btu/ft^3) and, thus, displace air which would normally be compressed in the compressor section of the gas turbine. The gas turbine would then operate at higher efficiency because there would be less compression work for the same net power output. The incentive to produce a clean, gasified fuel suitable for gas turbines is quite high since the use of such a fuel would allow the operation of gas turbine central stations which should be less costly than comparable steam stations and should operate at efficiencies equal to or better than those envisioned for steam power systems.

COGAS Systems

A second and potentially more promising system utilizing gas turbines is the combined gas and steam (COGAS) power system. A simplified schematic diagram for an exhaust-fired type of COGAS system is shown in Fig. 11. Fuel from the gasification process would be fed into the burner of a high-temperature gas turbine. After combustion and expansion through the gas turbine, the hot combustion products would be exhausted into a heat recovery boiler to raise steam for expansion through a steam turbine. Supplementary firing in the boiler would be optional. The application of COGAS power systems to large-capacity, base-load electric power generation (Refs. 15 and 16) has been limited primarily to the US Southwest where large quantities of low-cost natural gas are available. Even in this area, the small improvements in performance and cost offered by present-day COGAS systems relative to those of the conventional steam stations have not been sufficiently high to induce utilities to convert from conventional steam to COGAS systems. In those few large COGAS systems that have been put in operation, supplementary firing is employed in the boiler and the gas turbines produce less than 20% of the total station output.

Early results of this investigation indicated that station efficiency could be increased significantly if the amount of gas turbine participation were increased by reducing the amount of supplementary firing in the boiler. Further increases in COGAS performance would be possible by increasing the gas turbine inlet temperature. These trends are shown in Fig. 12 for methane-fueled COGAS systems incorporating a low-performance steam cycle (thermal efficiency of 34%) and gas turbine inlet temperatures of 2000, 2400, and 2800 F. The data in Fig. 12 clearly indicate that the best COGAS performance would be obtained if the steam boiler were of the simple waste-heat recovery type with no supplementary combustion. Detailed performance and economic analyses of various steam cycles for use in COGAS systems were carried out, resulting in selection of a 2400 psig/1000 F/1000 F steam cycle without feedwater heating. Performance estimates of COGAS systems using this steam cycle are shown in Fig. 13 for turbine inlet temperatures of 2000 F, 2400 F, and 2800 F, and for a range of applicable pressure ratios. These estimates are based on the use of both methane and a 162-Btu/ft^3 gas supplied at burner pressure. As with the simple gas turbine system, both cycle efficiency and power output per unit air-flow would be higher for the system burning low-Btu gas. The projected net station efficiencies of 50%, in systems using current technology, to 56% or 57% in later generations would be significant improvements over the efficiencies that might be realized by any other system except the very exotic and very expensive liquid-metal topping cycles. The COGAS system, however, would use machinery which is evolutionary in nature, i.e., machinery which is based upon actual power systems now being manufacgured.

By utilizing advanced cooling techniques such as precooled bleed air, the maximum turbine inlet temperature for the three time periods of interest are projected to increase to 2200, 2800, and 3100 F, respectively, resulting in efficiencies several points higher than those depicted in Fig. 13. The projected efficiencies of COGAS systems using precooled bleed air and burning low-Btu gas supplied at burner pressure and 150 F are shown in Fig. 14 to approach 58%, a value which is nearly 50% greater than now realized in conventional steam stations. This performance may be improved even more if the fuel gas were to be supplied to the system at a temperature higher than 150 F, as shown in Fig. 15 for a third-generation or 3100-F turbine inlet temperature system. At fuel gas temperatures of 600 F and above, the performance of the system could be boosted to values of 60% and over, a goal which should supply a tremendous incentive to fuel cleanup processes.

Preliminary estimates for the overall cost of electricity generated by a conventional steam system, a straight gas turbine system, and a GOGAS system are shown in Fig. 16. The projections presented in Fig. 16 demonstrate that the use of advanced technology in gas turbines could result in power systems which may produce electricity at costs equal to or even less than now realized from conventional steam systems, and still reduce the mission of sulfur oxides into the atmosphere. A second benefit arising from these systems is a reduction in thermal pollution of cooling water. The straight gas turbine rejects heat directly to the atmosphere; thus, there is no thermal pollution of cooling water. A reduction of thermal pollution by about 50% (compared to conventional steam stations) is possible with COGAS systems because of the increase in cycle efficiency, and because of the higher sensible heat content of the stack gases.

CONCLUSIONS

In conclusion, it can be said that the use of aircraft technology in industrial gas turbines may result in power systems which could produce electric power at reasonable cost using fuels which are appreciably more expensive than those used today, but which do not contain sulfur. The premise that advanced-cycle power systems could maintain the cost of producing electricity at levels now obtained with conventional systems has been shown to have great promise in future power systems. Additional benefits will occur through the use of advance-cycle power systems in areas of thermal pollution and in capital costs.

FUTURE WORK

Having determined the most promising generic classification of power systems, there remains a good deal of work to be done. Before detailed design work which would lead to actual engine development can be undertaken, further studies must be made with the objective of determining the best cycle configuration and operating conditions. Of immediate interest is the problem of combustion of the low-Btu fuel gas in the vitiated conditions occurring in a reheat combustor. The use of reheat in a COGAS cycle may allow significant gains in performance, but current limitations of funding do not allow a thorough study of this cycle. Operation of advanced-cycle power systems during transient periods will also require more detailed analyses, particularly in combination with the fuel gasification systems. Evaluation of the effect of these areas on system costs must also be made. Comparable work in the fuel clean up processes must also be performed, particularly in the area of high-temperature, high-pressure desulfurization techniques.

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TABLE I

ADVANCED-CYCLE POWER SYSTEMS INVESTIGATED

External-Combustion Systems

- Conventional Steam
- Binary Cycle Bottoming
- Binary Cycle Topping
- Closed-Cycle Gas Turbine

Internal-Combustion Systems

- Open-Cycle Gas Turbine
- Combined Gas and Steam (COGAS) Turbine

TABLE II

SUMMARY OF PERFORMANCE, AND CAPITAL AND GENERATING COSTS FOR CONVENTIONAL STEAM-ELECTRIC STATIONS

Fuel	No. Units and Size - Mw	Steam Conditions	Net Station Efficiency[1]	Capital Cost \$/kw[2]
		PRESENT-DAY		
Untreated Coal	2-500	2400 psia/1000 F/1000 F	36.6%	\$176.9
Gasified Coal	2-500	2400 psia/1000 F/1000 F	37.5	148.0
Untreated Oil	2-500	2400 psia/1000 F/1000 F	37.0	163.5
Desulfurized Oil	2-500	2400 psia/1000 F/1000 F	37.5	163.5
		SECOND-GENERATION		
Untreated Coal	1-1000	3500 psia/1000 F/1000 F	38.6	\$162.7
Gasified Coal	1-1000	3500 psia/1000 F/1000 F	39.5	136.9
Untreated Oil	1-1000	3500 psia/1000 F/1000 F	39.0	150.1
Desulfurized Oil[3]	1-1000	3500 psia/1000 F/1000 F	39.3	150.1
		THIRD-GENERATION		
Untreated Coal	1-1000	3500 psia/1000 F/1000 F	39.6	\$162.7
Gasified Coal	1-1000	3500 psia/1000 F/1000 F	40.5	136.9
Untreated Oil	1-1000	3500 psia/1000 F/1000 F	39.5	150.1
Desulfurized Oil	1-1000	3500 psia/1000 F/1000 F	40.5	136.9

(1) At a 70% annual load factor
(2) This is the installed cost for a station needing no unusual site preparation or station aesthetics.
(3) Residual oil may be gasified to meet second-generation requirements; thus the performance, capital costs, and generating costs for gasified oil-fired stations will be similar to those for the gasified coal station.

FIG. 1

AMMONIA BOTTOMING CYCLE

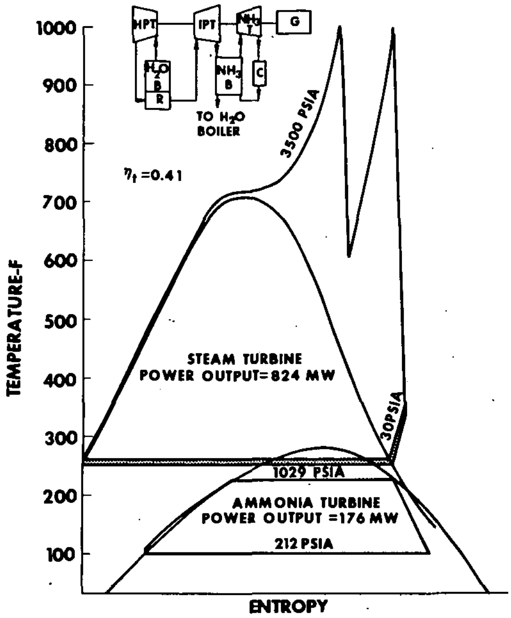

FIG. 2

HIGH-PERFORMANCE POTASSIUM TOPPING CYCLE

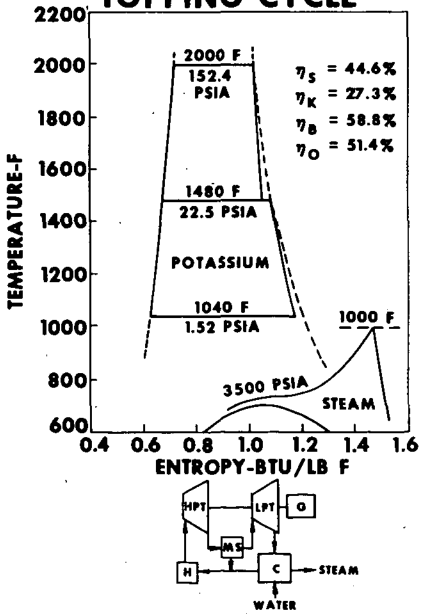

FIG. 3

HELIUM CLOSED-CYCLE TURBINE PERFORMANCE

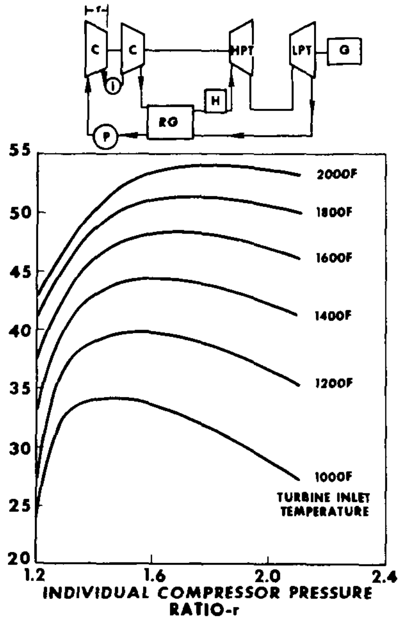

FIG. 4

CARBON DIOXIDE POWER CYCLE

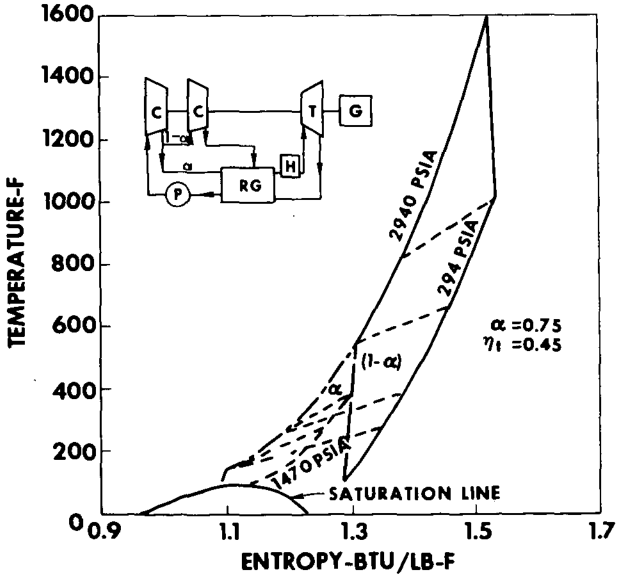

FIG. 5

HIGH PERFORMANCE SO_2 POWER CYCLE

REGENERATION EFFECTIVENESS = 92%
CYCLE THERMAL EFFICIENCY = 58%

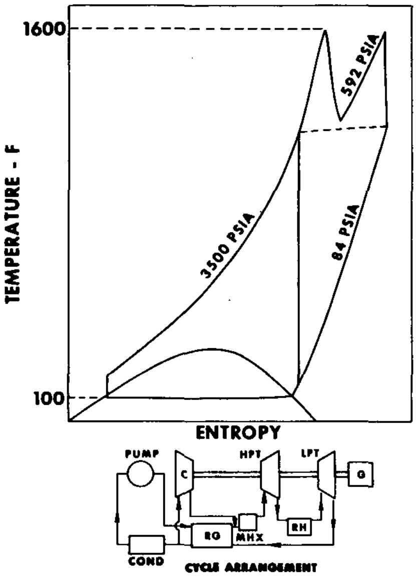

FIG. 6

COMPARATIVE ENERGY COSTS

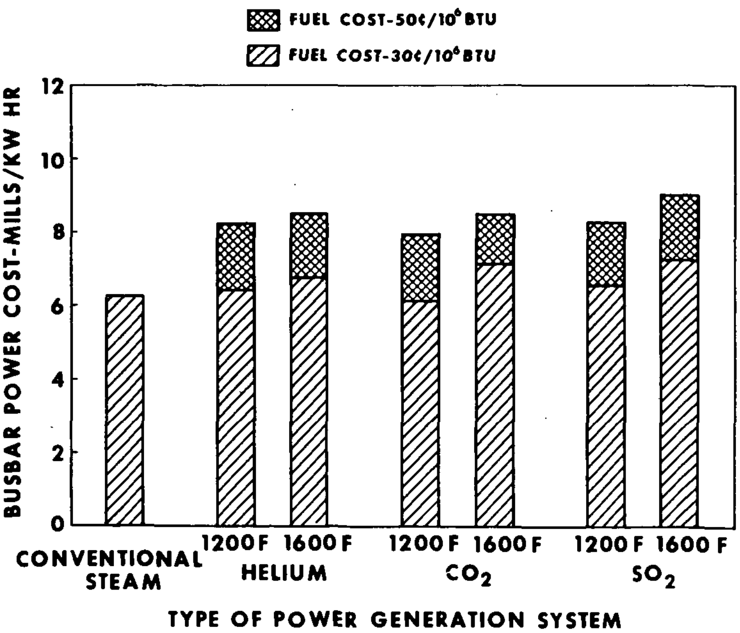

FIG. 7

PROJECTED DESIGN TECHNOLOGY FOR BASE-LOAD GAS TURBINE SYSTEMS

FUELS: METHANE (HHV = 1000 BTU/FT^3)
PRODUCER GAS (HHV = 162 BTU/FT^3)

PARAMETER	TIMF PERIOD		
	FIRST GENERATION (1970's)	SECOND GENERATION (1980's)	THIRD GENERATION (1990's)
TURBINE INLET GAS TEMPERATURE, F	1600 TO 2400	2000 TO 2800	2400 TO 3100
COMPRESSOR PRESSURE RATIO	8 TO 28	8 TO 36	8 TO 36
COMPRESSOR POLYTROPIC EFFICIENCY, %	89	92	93
TURBINE NOMINAL ADIABATIC EFFICIENCY, %	90	92	93
REGENERATOR AIRSIDE EFFECTIVENESS, %	60 TO 90	60 TO 90	60 TO 90
REGENERATOR TOTAL PRESSURE DROP, %	4 TO 8	4 TO 8	4 TO 8
TURBINE COOLING TECHNIQUE	IMPINGEMENT-CONVECTION	IMPINGEMENT-CONVECTION	1. IMPINGEMENT-CONVECTION 2. TRANSPIRATION
TURBINE COOLING AIR TEMPERATURE,F	1. SAME AS COMPRESSOR DISCHARGE AIR TEMPERATURE 2. PRECOOLED TO 125-TO-250 F		

FIG. 8

TURBINE INLET TEMPERATURE PROGRESSION FOR BASE-LOAD OPERATION

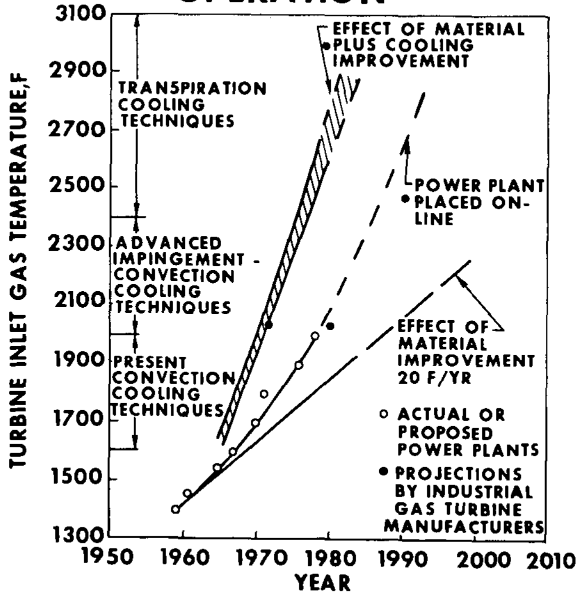

FIG. 9

EFFECT OF TURBINE COOLING FLOW ON BASE-LOAD SIMPLE-CYCLE GAS TURBINE PERFORMANCE

FUEL-METHANE (HHV = 1000 BTU/FT^3)
AMBIENT 80 F AND 1000 FT
FUEL DELIVERED TO BURNER AT COMPRESSOR OUTLET PRESSURE

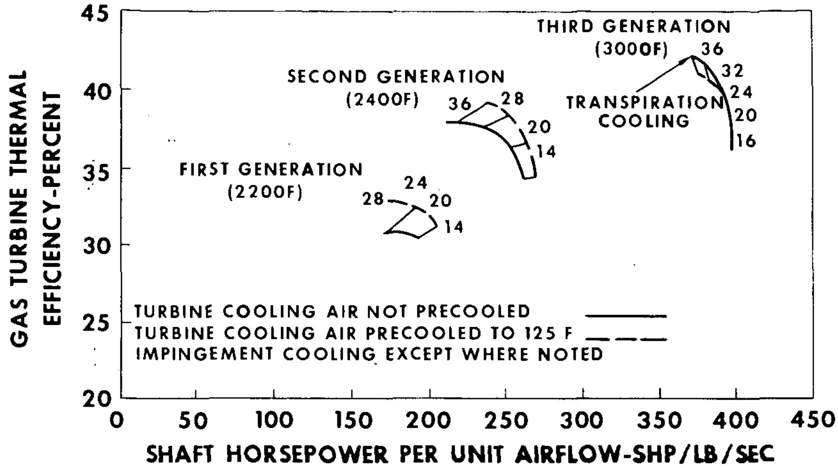

FIG. 10

THIRD GENERATION BASE-LOAD SIMPLE-CYCLE GAS TURBINE PERFORMANCE

FUEL-PRODUCER GAS (HHV = 162 BTU/FT^3)
AMBIENT-80 F AND 1000 FT
TURBINE COOLING CONFIGURATION :
ADVANCED IMPINGEMENT - CONVECTION

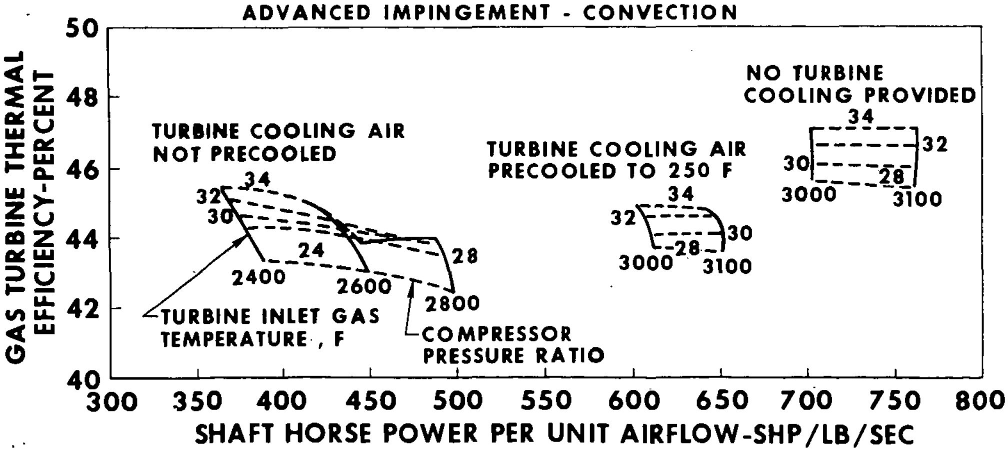

FIG. 11

COMBINED GAS-STEAM TURBINE SYSTEM

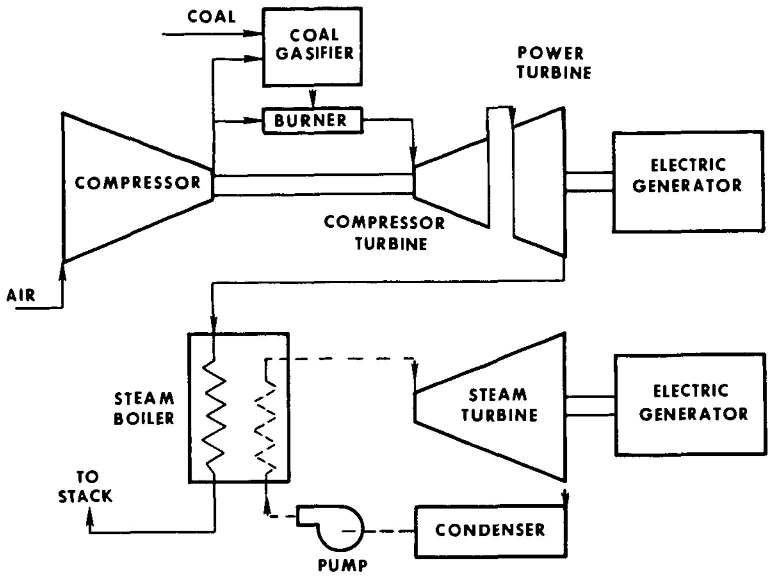

FIG. 12

PERFORMANCE OF EXHAUST-FIRED COMBINED SYSTEM

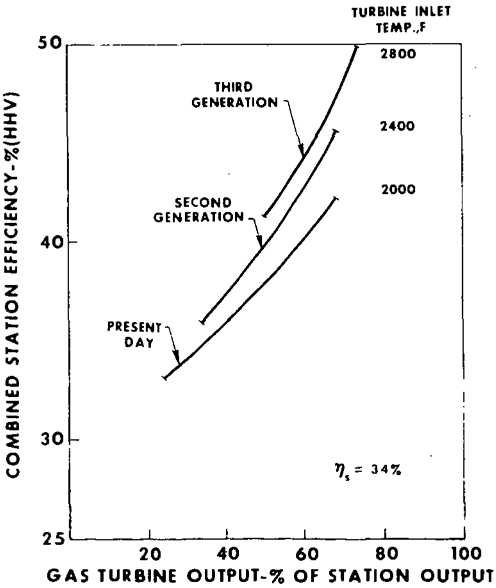

FIG. 13

PERFORMANCE OF UNFIRED WASTE HEAT COMBINED CYCLE

BASED ON USE OF 2400 PSIG/1000F/1000F STEAM CYCLE
EFFICIENCY = 38.8%
——— METHANE SUPPLIED AT BURNER PRESSURE AND 80F
– – – – 162 BTU/FT³ GAS SUPPLIED AT BURNER PRESSURE AND 150F

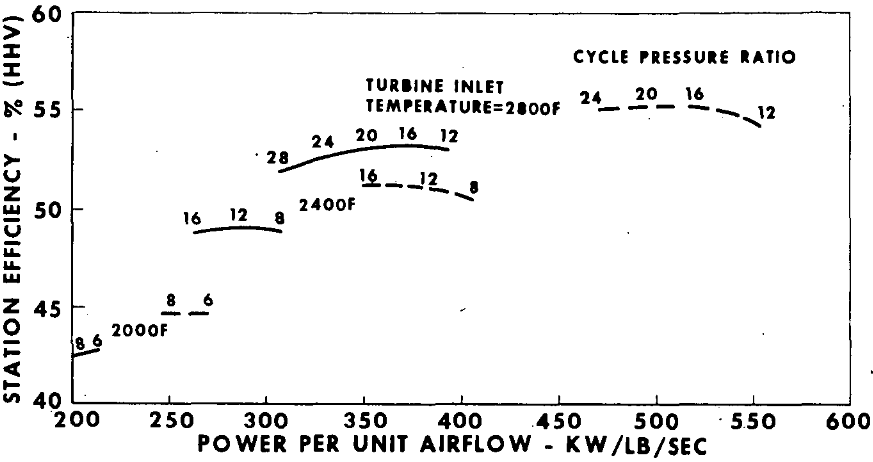

FIG. 14

PERFORMANCE OF UNFIRED WASTE-HEAT COMBINED CYCLE

FUEL - 162 BTU/FT^3 GAS SUPPLIED AT BURNER PRESSURE AND 150 F
STEAM CYCLE - 2400 PSIG/1000 F/1000 F, EFFICIENCY = 38.8%

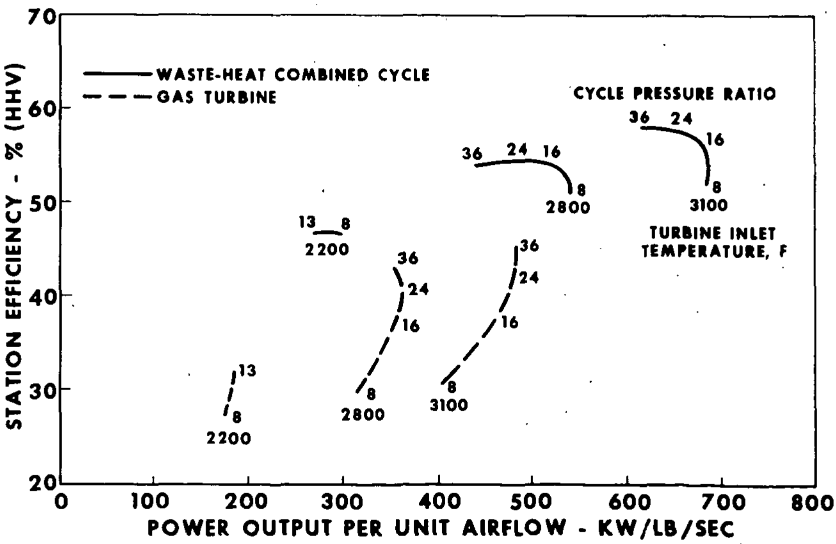

FIG. 15

EFFECT OF FUEL SUPPLY TEMPERATURE ON STATION PERFORMANCE

FUEL-162 BTU/FT^3 GAS SUPPLIED AT BURNER PRESSURE
STEAM CYCLE-2400 PSIG/1000 F/1000 F, EFFICIENCY = 38.8%
TURBINE INLET TEMPERATURE = 3100 F
CYCLE PRESSURE RATIO = 24

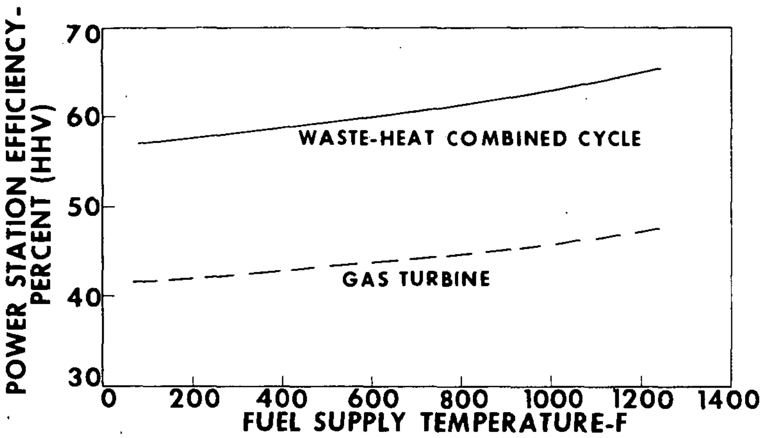

FIG. 16

BUSBAR POWER COSTS

1000-MW STATION
70% ANNUAL LOAD FACTOR

$$\text{POWER COST} = \frac{\text{CAPITAL CHARGES + FUEL COST + LABOR + MAINTENANCE}}{\text{KW-HR/YR}}$$

$$\text{FUEL COST} = \frac{\text{BASE FUEL COST}}{\text{STATION EFFICIENCY}}\ ¢/10^6\ \text{BTU}$$